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Impact Testing of a Stirling Convertor's Linear Alternator

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Abstract. The U.S. Department of Energy (DOE), in conjunction with NASA John H. Glenn Research Center and Stirling Technology Company, are currently developing a Stirling convertor for a Stirling Radioisotope Generator (SRG). NASA Headquarters and DOE have identified the SRG for potential use as an advanced spacecraft power system for future NASA deep-space and Mars surface missions. Low-level dynamic impact tests were conducted at NASA Glenn Research Center's Structural Dynamics Laboratory as part of the development of this technology. The purpose of this test was to identify dynamic structural characteristics of the Stirling Technology Demonstration Convertor (TDC). This paper addresses the test setup, procedure and results of the impact testing conducted on the Stirling TDC in May 2001.

INTRODUCTION

The U.S. Department of Energy (DOE), in conjunction with NASA John H. Glenn Research Center (GRC) and Stirling Technology Company (STC) are currently developing a Stirling convertor for use in an advanced radioisotope power system to provide spacecraft on-board electric power for NASA's deep-space missions and for Mars surface rovers. STC of Kennewick, WA is under contract to DOE to develop a radioisotope Stirling convertor. NASA GRC is providing technical consultation for this effort based on their expertise in Stirling technologies dating back to the mid-1970's.

Stirling is being evaluated as an alternative replacement to Radioisotope Thermoelectric Generators (RTGs). Due to the Stirling system's efficiency (over 20 percent), just one-third the amount of Plutonium is required compared to the RTGs, thereby significantly reducing fuel cost (Schreiber, 2001).

Multiple Stirling units have demonstrated Stirling Technology Demonstration Convertor (TDC) power, efficiency and long life. One of STC's terrestrial radioisotope Stirling convertor has operated maintenance-free and without performance degradation for over 65,000 hours (7.4 years). Such long lifetimes are required for deep-space missions to the outer planets where solar power is not an option.

In preparation for possible deep space missions, an operating (power-producing) 55-We Technology Demonstration Convertor was dynamically tested in December 1999 at NASA GRC's Structural Dynamics Lab (SDL) (Goodnight, 2000; Hughes, 2000). This TDC was tested to levels beyond those used to vibration qualify the comparable RTGs used on the Cassini Mission. Subsequent emissions testing of two TDC's in tandem were also performed to characterize the structure borne disturbances produced by the TDC, and thereby establish vibratory compatibility requirements for possible nearby scientific hardware. The effects of Stirling power package dynamics were studied in both the random qualification and vibratory emissions testing.

TDC design robustness was demonstrated in the December 1999 vibration qualification testing, although a clear understanding of what contributed to this result remained unknown. The physical construction of the TDC in its multi-layered configuration is a primary source of the TDC's robustness. The mounting configuration may also have contributed. The linear alternator is the heaviest component of the TDC and is overhung off the piston housing as shown in Figure 1. The large diameter flange of the piston housing was mounted to the rigid fixture in the December 1999 vibration qualification test (and also in the May 2001 impact test). As part of the understanding of the dynamics of the Stirling TDC, particularly the susceptibility of its linear alternator to the random launch environment, an in-situ modal survey of a motored TDC (Unit #5) was performed, in May 2001.

Both random baseshake and impact testing were done. The low-level hammer impact tests are reported in this paper. The linear alternator's pressure vessel was removed during the impact testing to allow physical access to the piston/mover rod. The TDC's natural frequency is 71 Hz and is in fact lowered from its operational frequency (79-82.5Hz) due to missing internal gas. The alternator acted as a linear motor during the random baseshake tests, thereby reversing the usual power producing cycle. During the impact tests, the TDC was not operational or producing power.

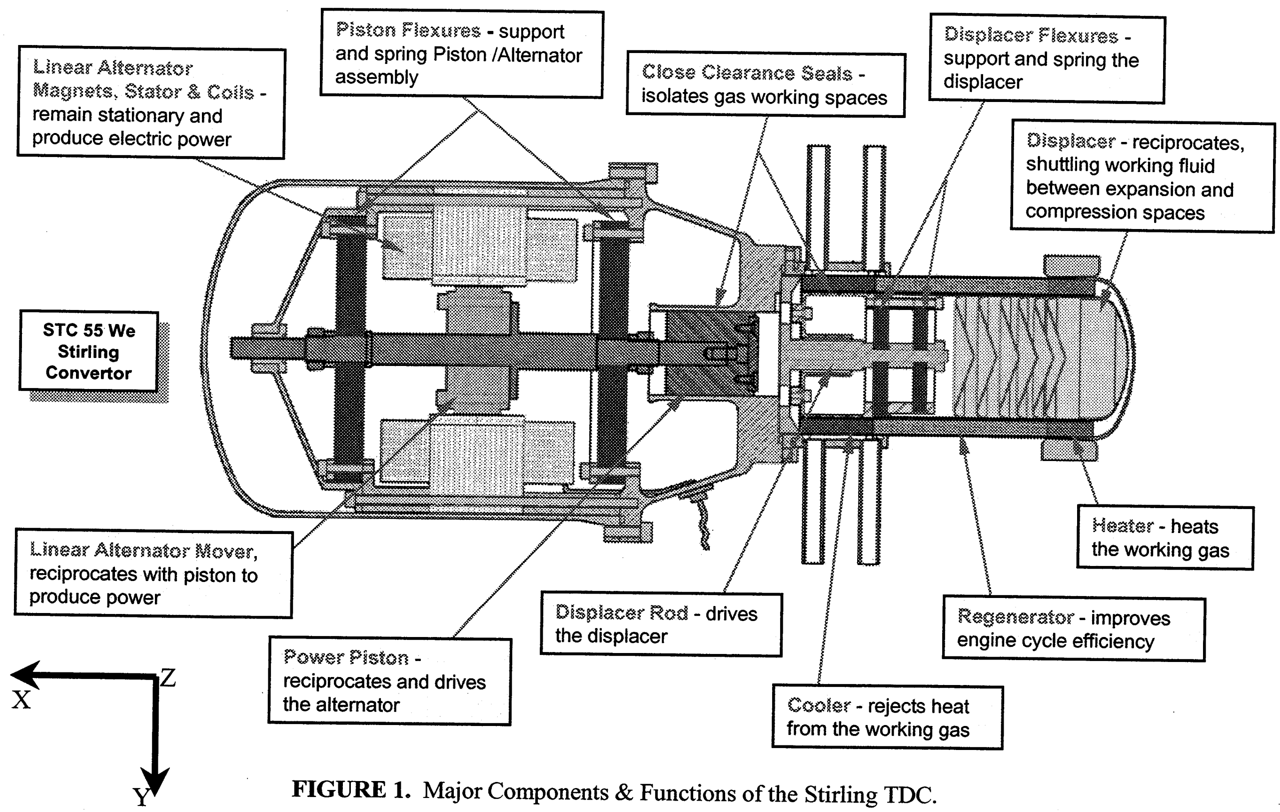

FIGURE 1. Major Components & Functions of the Stirling TDC.

TEST OBJECTIVE

As mentioned previously, an early TDC was exposed to non-deterministic random input. Even though the objectives from the vibration test were met and exceeded, some decreases in power at maximum excitation levels were noticed during the lateral testing (Y-axis) specifically at qualification levels (Goodnight, 2000; Hughes 2000) (see Figure 2). The TDC returned to full power after the qualification levels were removed. In May 2001, impact tests were performed on the Stirling TDC at NASA GRC's SDL. The objective of the impact test was to identify dynamic structural characteristics of the Stirling Technology Demonstration Convertor (TDC) and specifically to determine the predominant frequencies of the TDC in the plateau region (50 to 250 Hz) of the Design/Qualification specification.

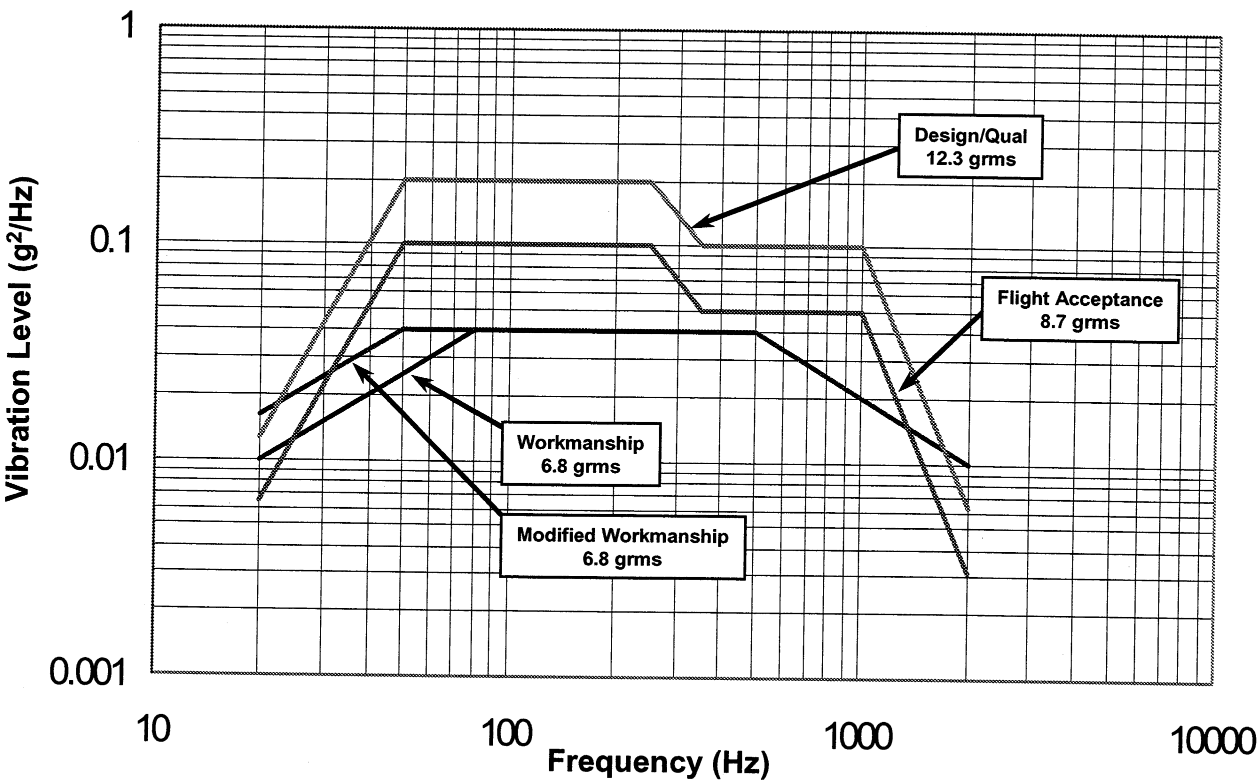

FIGURE 2. Stirling TDC Random Vibration Test Levels.

TEST SETUP

The Stirling TDC was supported in a specially made test fixture as shown in Figure 3. The test fixture was rigidly mounted to the 60 inch by 72-inch slip table (~500 lbs. without test article) similar to the vibration qualification configuration as shown in Figure 4.

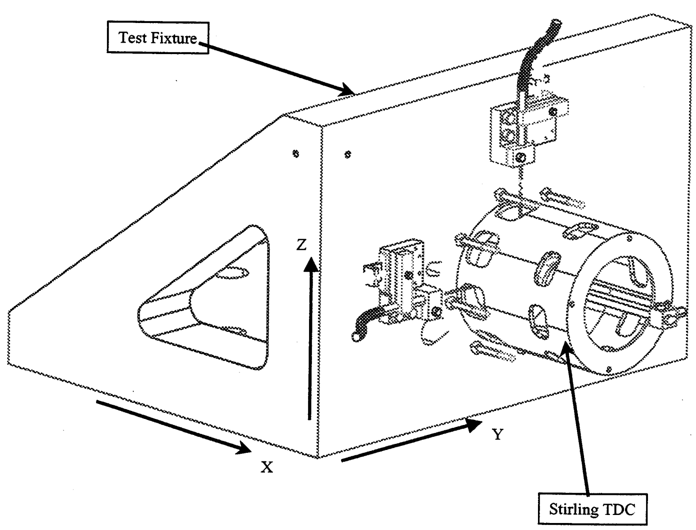

FIGURE 3. Alternator End of Stirling TDC.

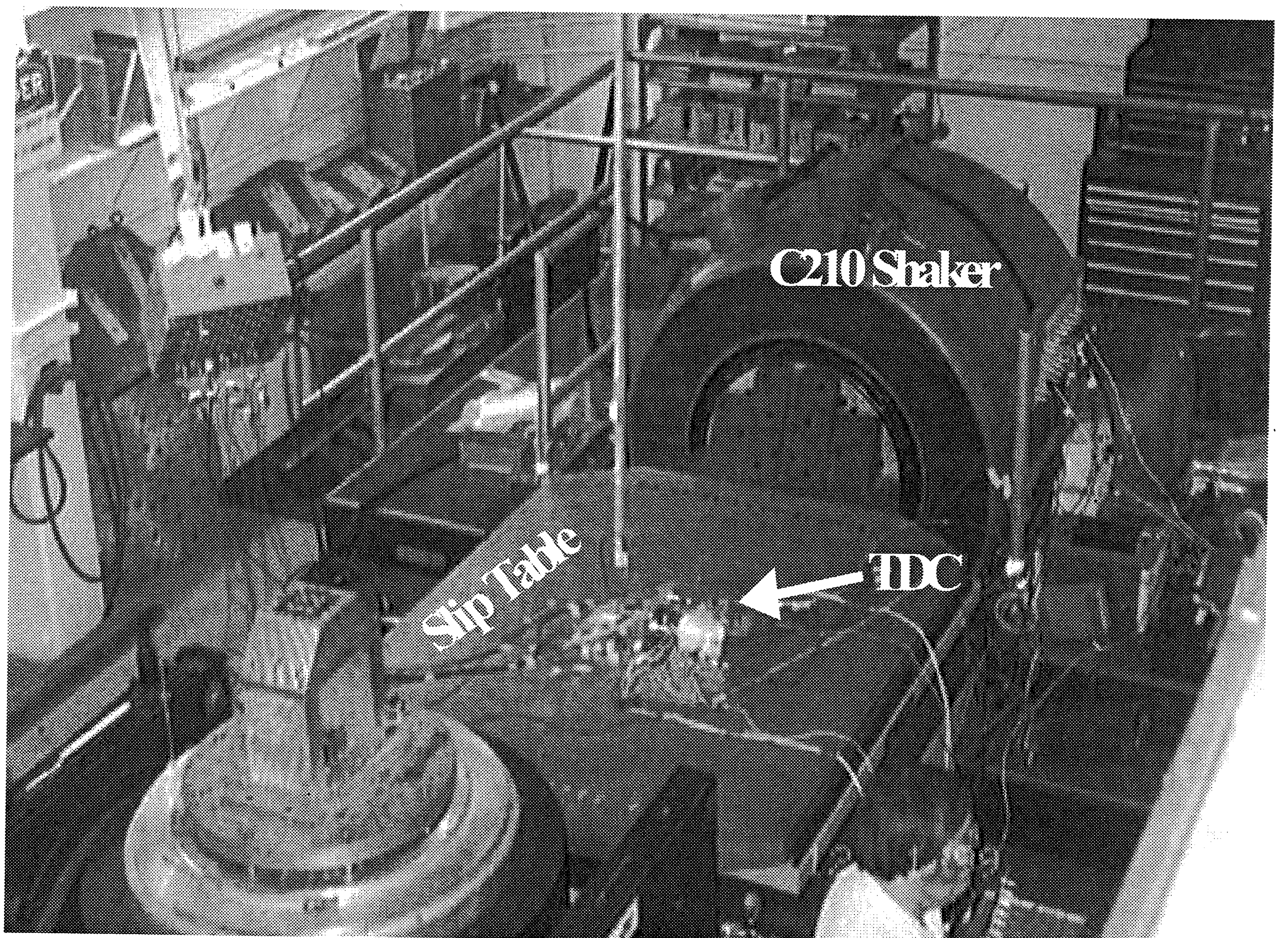

FIGURE 4. Overview of the Vibration Test Setup.

A total of eleven accelerometers (Endevco 22, 23 and 2222) were placed on the Stirling TDC and the fixture (see Table 1). Nine accelerometers were selected to capture the dynamic behavior of the Stirling TDC and the remaining two accelerometers were mounted on the fixture to capture the fixture's response to the impact. A PCB instrumented hammer with a soft tip was utilized to induce energy into the TDC. The impact test data was stored in the time domain and immediately reviewed for data validation.

TABLE 1. Location of the Accelerometers.

Accelerometer Label	Accelerometer Location
2X- 2Y- 2Z-	Outboard End of Piston/Mover Rod
3Y+ 3Z-	On Piston/Mover Rod outboard flexure
4Y+ 4Z-	On Piston/Mover Rod inboard flexure
5X-	Mid-span outboard Alternator Flexure
6Y-	Flexure base/stator/casing
11X+ 11Y-	Test fixture

TEST DESCRIPTION

The impact testing started in the axial axis (X, direction of the TDC's piston stroke), followed by the lateral (Y, perpendicular to the TDC's piston stroke and parallel to the slip plate). The test was completed with impacts on the vertical axis (Z, perpendicular to the slip plate). The locations where the Stirling TDC was impacted are shown in Figure 5. The TDC was impacted at the end of the piston/mover rod in the X and Y-axes. Other impacts were also performed on the aft bearing assembly in all axes. All impact data was reviewed prior to proceeding to the next impact.

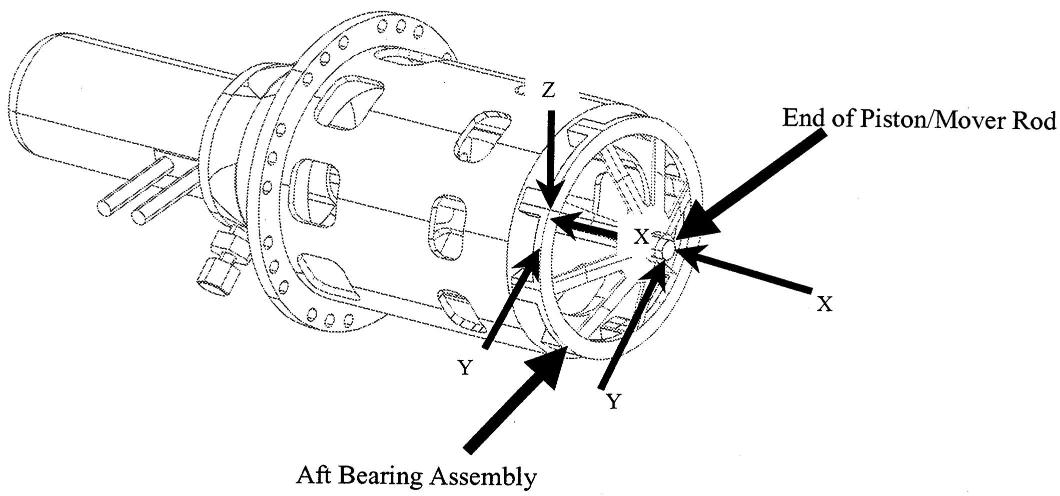

FIGURE 5. Impact Locations.

DATA PROCESSING

Different approaches were applied during the data processing. All data was collected and post processed using a HP VXI data acquisition system and IDEAS-8 Master Series 8M3 software. In the final analysis, a bin width of 0.625 Hz was used with no window applied. The windowing approach was appropriate as the data was near zero at the beginning and end of each sample.

The H_1 Frequency Response Function (FRF) method (Equation 1) was used. This formulation, described below, minimizes the noise of the output. It is susceptible to noise on the input and underestimates the analytical system's frequency response function. In this test the only input was the hammer and it was relative easy to minimize the noise. Since the main objective was to identify major resonances (and not damping value or amplitude) H_1 was utilized to fulfill the objective.

$$H_1 = \frac{S_y \times S_x^*}{S_x \times S_x^*} = \frac{G_{yx}}{G_{xx}} \tag{1}$$

Where,

S_y (f) – linear Fourier spectrum of y(t) (time domain output of the system)
S_x (f) – linear Fourier spectrum of x(t) (time domain input of the system)
G_{yx}(f) – cross spectrum of y(t) and x(t) – complex valued
G_{xx}(f) – autopower spectrum of x(t)
* – indicates complex conjugates

Part of the processing involved the development of a simplified model of the basic geometry of the test unit to visualize the mode shapes obtained from the FRF curve fitting analysis. This Test Empirical Model (TEM) (see Table 2 and Figure 6) of the Stirling TDC was used to visualize the modes and was based only on the number of accelerometers used during the test. A finite element model of the Stirling TDC was not available; therefore, a proper Test Analytical Model (TAM) was not developed.

TABLE 2. Stirling TDC Dynamic Traceline Model.

Traceline #	Represents	Nodes
1	Rod, Piston/Mover	2-4
2	Forward Flexure alone	3, 5-6
3	Alternator Stator/Casing	6, 11

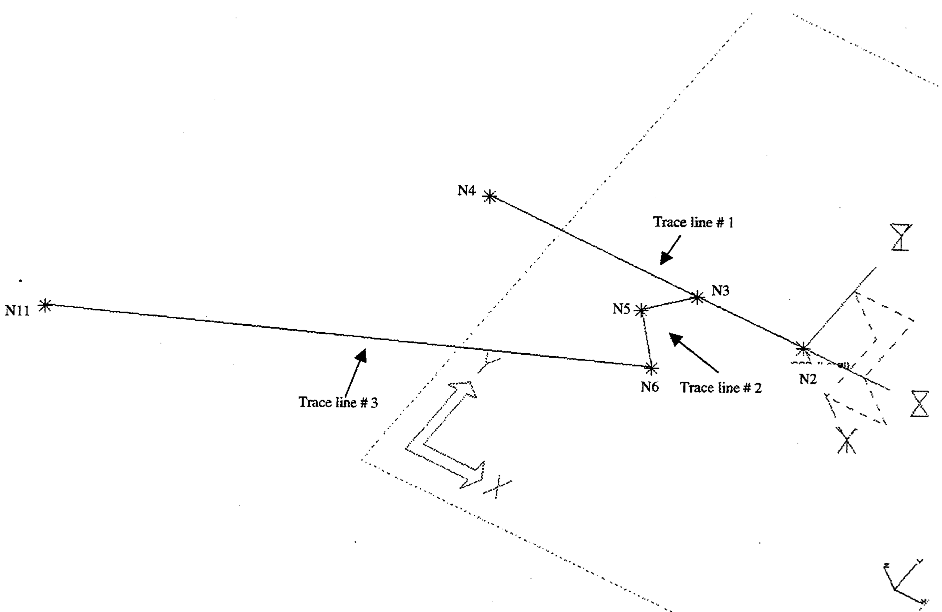

FIGURE 6. Test Empirical Model (TEM) of the Stirling TDC.

TEST RESULTS AND DISCUSSION

The impact testing of the Stirling TDC was conducted successfully. Several distinct classes of modes were observed. The first group directly involved the linear alternator's piston/mover rod. These were primarily the piston/mover rod response on its supporting springs/flexures. The second set of modes of the flexures alone, while important, were of secondary interest, and were identified as separate and distinct from the piston/mover rod modes. The final sets of modes were those associated with the displacer and linear alternator's casing/stator. These were of prime importance to the successful December 1999 vibration qualification test verification (Goodnight, 2000; Hughes, 2000), as all baseshake energy must pass through the piston housing and the linear alternator's casing and stator to reach the critical clearances which can affect the TDC's power performance.

The linear alternator's piston/mover rod displayed two fundamental responses. The initial response was that of the rod's flexural response superimposed on the primary drive frequency. The second observed modal response was the rod's response to the flexure's radial stiffness, which is much stiffer than the lateral stiffness. These phenomena were clearly seen in the rod's FRF (see Figure 7). This data showed the strongest response at the TDC's fundamental natural frequency of 71 Hz. This was a global phenomenon as all the accelerometers located on the rod were involved independent of the impact direction. With help of the empirical traceline model it was found that at 71 Hz the rod went through a combined axial (translational) and bending motion in both the Y and Z-axes. This behavior of the shaft at 71 Hz is believed to occur due to the interaction of the flexures on the shaft as it moves back and forth. This connection may have induced a moment causing the shaft to bend in the Y and Z-axes, as the shaft translated. The behavior of the 71 Hz mode may explain part of the reduction in power observed at high vibration test levels during the December 1999 vibration qualification test (Goodnight, 2000; Hughes 2000). The radial response of the rod against the radial stiffness of the flexure resulted in a pair of modes in the 400 Hz range. The slight differences in this modal pair are thought to be either a fixture stiffness effect or an effect of impact directionality.

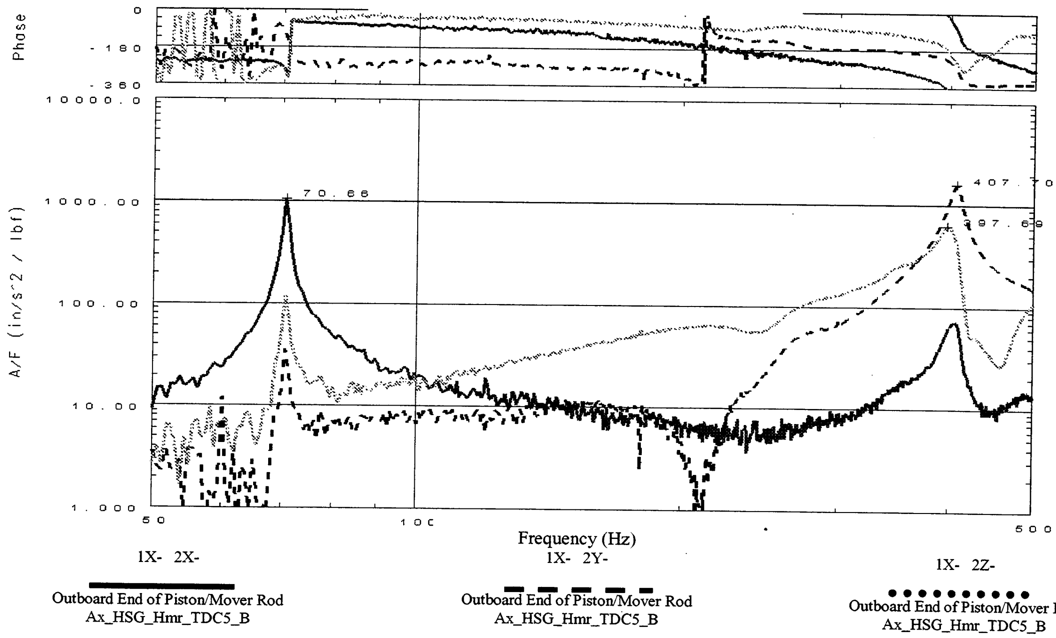

FIGURE 7. Piston/Mover Rod FRF.

The flexure alone response was also observed (see Figure 8). There was limited data (i.e. one accelerometer 5X-, as part of traceline 2). The out of plane response was observed. It is postulated the 523 Hz response is the symmetric flexure response, with all the spring(s) leaves flexing together in phase, and the 1253 Hz is the frequency of an anti-symmetric mode or higher order bending mode. A single accelerometer on the flexure itself does not allow determination of relative phasing or mode shape determination.

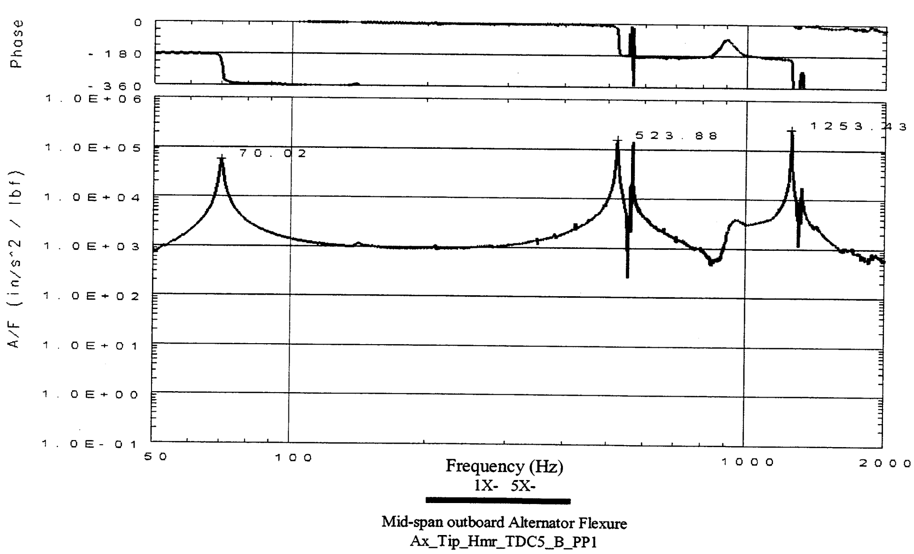

FIGURE 8. Flexure FRF.

All launch environments must be transferred through the piston housing and casing/stator to excite the piston mover/rod. Very interesting data was observed relating to this primary transmission path of external energy (i.e. through the launch interface or through the mounting flange during baseshake random testing). Impacts both to the piston mover/rod and to the aft bearing assembly (i.e. casing/stator) are shown in Figure 9. Clearly the modal response around 134 Hz is highly damped and typical poly-reference modal parameter techniques were unsuccessful in discerning this mode. The mode at 673 Hz was also significantly damped and easily recognizable (see Figure 9). Despite this clearer mode presentation, poly-reference techniques did not identify a good fit for this mode either. Stepping back to some more rudimentary methods, and based on the clear separation of these modes, a single degree of freedom (SDOF) circle fit was attempted. Circle fitting techniques revealed a clear frequency (134 Hz) and very high damping (29.4%) (see Figure 10). This damping result is not a material type damping and most likely is associated with the displacer on the opposite side of the mounting flange. This result cannot be confirmed, as the displacer was not instrumented because physical access was limited. The second mode observed at 673 Hz (6.03 % damping) is believed to be the primary casing/stator mode, as it is the highest peak in the FRF. The damping again is significant (see Figure 10), greater than material damping but not as high as the 134 Hz mode. Both of these natural frequencies are separated from the frequencies found for the rod and flexure response. These large damping values obviously contributed to the successful vibration qualification test verification in December 1999.

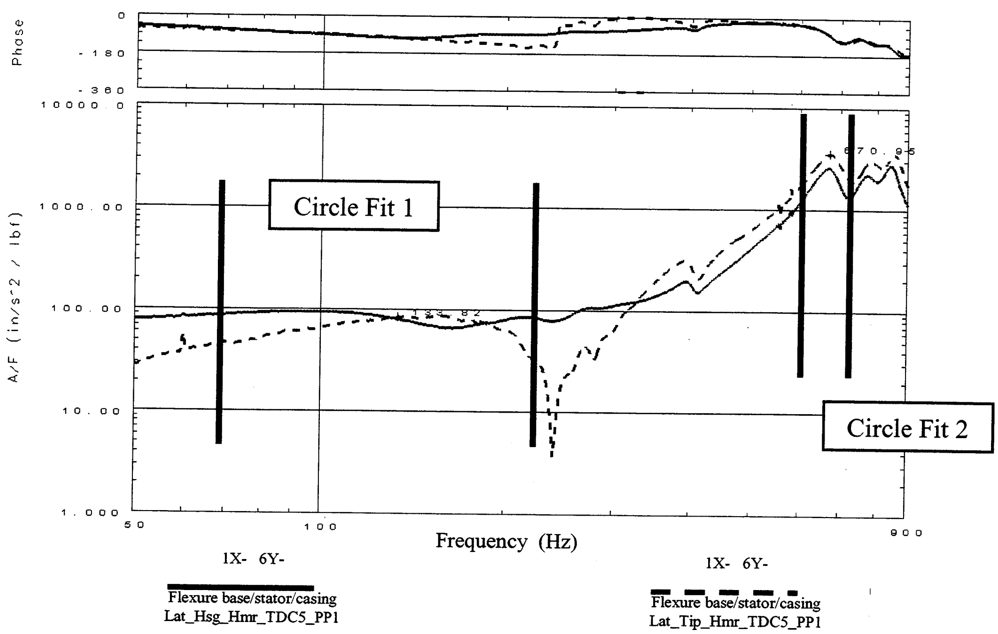

FIGURE 9. Casing/Base of Flexure FRF (Displacer and Stator Modes).

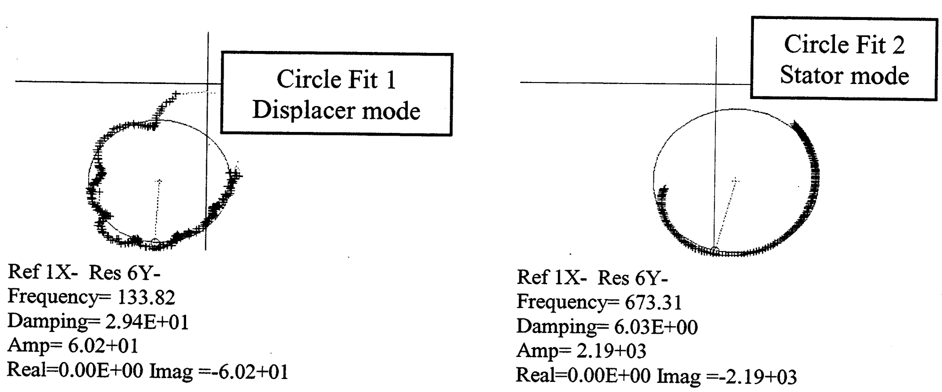

FIGURE 10. Displacer and Stator Argand Plots.

The aforementioned modes (see Table 3) are combined in a test vs. test Modal Assurance Criteria (MAC) Evaluation (see Figure 11). A high diagonal (~ 1.0) and low off-diagonal (< 0.2) MAC values are indicative of orthogonal (independent) modes. The results of this test are limited by the available instrumentation. For example, missing a mass matrix or under sampling spatially could lead to an erroneous conclusion of interrelated modes, where in actuality the sparse accelerometer distribution limits our insight into the modal data. A good example of this is the 134 Hz and 673 Hz modes (Figure 9). The circle fits clearly indicates two different modes (see Figure 10); however, there is no differentiation in the MAC data (Figure 11). This result is expected since only one accelerometer (6Y-) was located on the stator housing at the base of the outboard flexure outer ring. Another example is the frequency pair of 523 Hz and 1253 Hz (Figure 8).

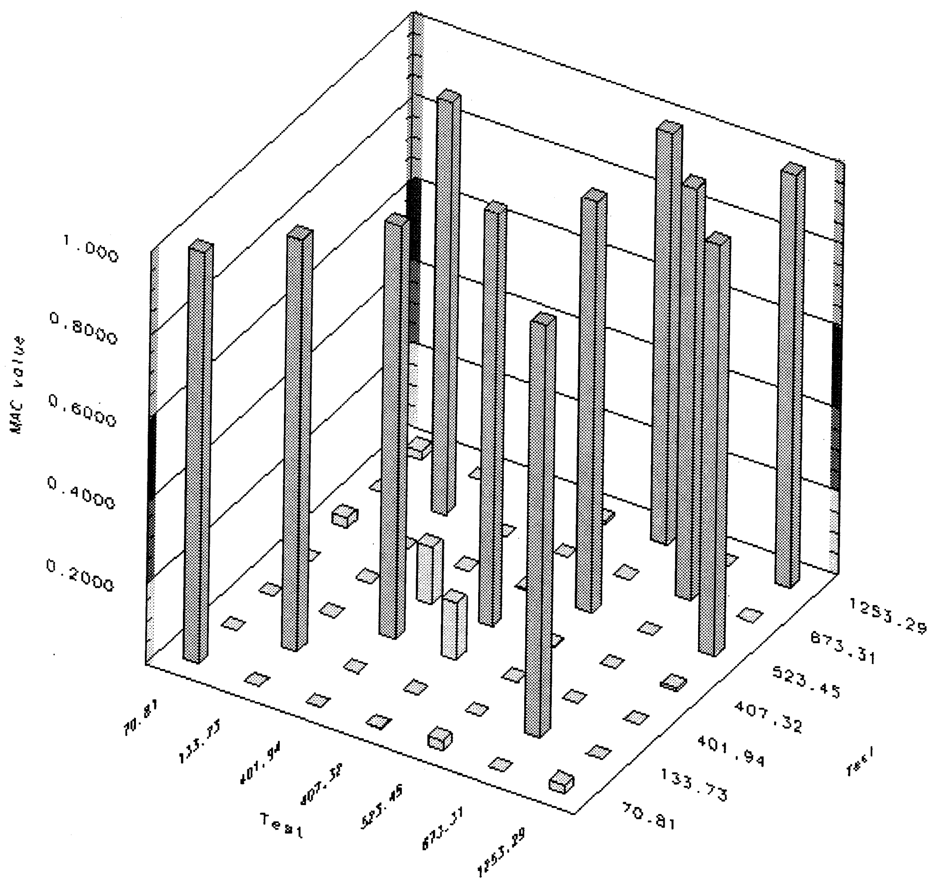

FIGURE 11. Modal Assurance Criteria Matrix.

TABLE 3. Stirling TDC Principal Modes.

Frequency Hz	Mode Description	% Critical Damping
71	Piston/Mover Rod combined axial and bending response in both Y and Z-axes	0.47
134	Casing/Displacer response	29.4
402	Piston/Mover Rod Z-axis cantilever/ radial response	0.17
407	Piston/Mover Rod Y-axis cantilever/ radial response	1.55
523	Symmetric Flexure bending response	0.28
673	Stator/Casing response	6.03
1253	Anti-symmetric Flexure bending response	0.11

CONCLUSIONS

In May 2001, impact tests were successfully performed on the Stirling TDC at NASA GRC's SDL. As a result of this test, resonances of various Stirling TDC components have been identified (see Table 3). The modal survey, focusing on the linear alternator side of the TDC, was limited by the number of accelerometers and the absence of an analytical Test Analysis Model. There were three primary areas of interest, the piston/mover rod, the flexure alone response, and the casing/stator response. The empirical TAM (the Test Empirical Model consisting of tracelines alone) was illustrative of the reported phenomena, but there is still a fair amount of conjecture in the interpretation of the results due to limits in the observability criterion.

Relative to the primary objectives of the test, the results helped to explain the success that the TDC enjoyed during the 1999 vibration qualification testing. There were only two modes of the TDC with frequencies in the primary plateau region (50 – 250 Hz of Figure 2); the 71 Hz piston/mover rod combined response, and the unnaturally highly damped 134 Hz mode observed in the casing/displacer. Since the 71 Hz mode is within this plateau region of the qualification level, it is believed that the combined flexural response with the axial response at this frequency may produce friction and leakage losses. This response phenomenon may provide a possible explanation of the observed drop of power during the lateral vibration testing at high vibration levels due to the loss of clearance in tight toleranced areas of the TDC. Separation of the bending and axial modes in the flight unit should provide higher tolerance at full power to launch loads. The high damping observed in the two casing related modes and the clear separation of the stator mode frequency from the operating frequency was also beneficial, and aids in the robustness of the TDC. The remaining modes found outside the initial plateau region (> 250 Hz) were significant in completing the description of the alternator response, however these modes were not excited to the same excitation levels.

In closing, these conclusions must be considered in light of the TDC's mounting configuration. The TDC was mounted by the large flange of the piston housing for our tests and the results would be affected by other choices of mounting configurations. Lastly, the implementation of system mounting frequency (i.e., where the multiple TDC's are combined into a single power unit) may have an effect on these results unless clearly decoupled (e.g., a fundamental system mounting frequency of 30-40 Hz).

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13. ABSTRACT ***(Maximum 200 words)***

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